*Research Article*

# Design and Stress Analysis for Spur Gears Using Solid Works Simulation

**M. Ranjith Kumar[1], S. Raja Narayanan[2], R. Dhinesh Kumar[3], S. Kandaguganesh[4], S. Muthusamy[5]**

[1]Assistant Professor, Department of Mechanical Engineering, M. Kumarasamy College of Engineering, Karur, Tamil Nadu, India.
[2]Assistant Professor, Department of Mechanical Engineering, M. Kumarasamy College of Engineering, Karur, Tamil Nadu, India.
[3]UG Student, Department of Mechanical Engineering, M. Kumarasamy College of Engineering, Karur, Tamil Nadu, India.
[4]UG Student, Department of Mechanical Engineering, M. Kumarasamy College of Engineering, Karur, Tamil Nadu, India.
[5]UG Student, Department of Mechanical Engineering, M. Kumarasamy College of Engineering, Karur, Tamil Nadu, India.

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**Abstract:** Spur gear is the most important component which is used to transfer the power and it also transmits the motion among the machine parts between two parallel axis. The reason for causing failures in Spur gear are compressive stress, friction, bending stress, wear and tear etc. In this study the compressive stress and bending stress of a spur gear are calculated mathematically with the help of PSG DDB, and the spur gear model is simulated by using solid works software. The PSG DDB values and formulae's are used in the mathematical calculations and then compared with solid works analysis. The final out comes are checked by comparing those results and we choose a better material for our assumed data. (PSG DDB –PSG Design Data Book)

**Keywords:** Spur Gear, Stresses, Solid Works, Mesh, Analysis.

## 1. Introduction

One of the most crucial mechanism in automotive industry and rotary machines are meshing the gears and it is considered as the best method for power transmission.

The gear material is selected from PSG DDB are Cast Iron (Grade 35), C45 Steel, Steel (15Ni2Cr1Mo15) for designing of spur gear. There different types of parameters are required to select the gear materials like ultimate tensile stress, Minimum endurance limit stress for complete reverse of stresses (kgf/cm$^2$), Design surface stress and Design bending stress.

One of the most important properties of the gear tooth is that the contact point is constant motion along with the face of the gear tooth.

The rapid growth in the industrial revolution, especially in the machineries and automotive industry, that have been pushed the design engineers to increase the usage of gears in their design in the past years.

During the meshing operation the stress is distributed along the gear profile and the compressive stress takes place at the time of meshing after that bending stress takes place.

All the three material's stresses are calculated manually and those values are compared with simulated value to choose the better material for our assumed data.

The three selected materials have different Young's modulus and poison's ratio so the hardness of the material vary with each other. Each material have some properties to carry the load on their tooth, in this study we clearly understand which material can easily carry the load without any failure.

The power transmission mechanism in all automotive industries and machineries are rapidly increased in the past years. The gears design in the automobile industry and generators are mainly required more precise checking to ensure the maximum safety for gears.

The most important test after the manufacturing of gears is the distribution of stress, especially compressive stress, on the face of the gear teeth.

The stress analyses of the spur gears by using mathematical calculations, which have some assumptions then compared that data with solid works software analyzed value.

In this case, the bending and compressive stress are analyzed, and then designed the spur gears in solid works for software analyzed values. And the stress limit is taken from PSG DDB, if any one of the value is higher than stress limit value that material is neglected and the design is considered as unsafe.

The compressive stress and bending stress between the wheel and pinion can be calculated by using PSG DDB formulae's. All the values calculated are in kgf/cm$^2$.

## 2. Parameters to be Analysed

- The modelling of gear is done by using SolidWorks 2019 software.
- The tooth forms of spur gear are standard PSG DDB full depth involute profiles.
- Finally, all results are compared and we are going to decide which material is best for that particular calculation.

## 3. Data Used for Calculation

- Power to be transmitted = 22500w.
- Speed = 900 rpm.
- Velocity ratio= 2.5:1.
- Pressure angle = 20 degrees.
- Life = 10,000 hours.
- We are going to design a spur gear for the following specifications.

## 4. Gear Materials

- The materials used in designing of spur gear are Cast Iron (Grade 35), C45 Steel, Steel (15Ni2Cr1Mo15).
- The materials chosen are linear isotropic in nature and does not exhibit any change in their own properties. Mostly Stainless Steel is generally used in making of spur gears.
- Following represents the material properties of Cast Iron (Grade 35), C45 Steel, Steel (15Ni2Cr1Mo15) respectively.
- All the Material Properties are taken from PSG DDB.

### 4.1 Material Properties of Cast iron (Grade 35)

**Table 1**

| Sl No | Description | Value | Unit |
|---|---|---|---|
| **1** | Tensile strength | >=35 | kgf/mm2 |
| **2** | Limit Stress | 16 | kgf/mm2 |
| **3** | Poisson Ratio | 0.28 | |
| **4** | Surface stress | 7500 | kgf/cm2 |
| **5** | Bending stress | 750 | kgf/cm2 |

### 4.2 Material Properties of C45 Steel

**Table 2**

| Sl No | Description | Value | Unit |
|---|---|---|---|
| **1** | Tensile strength | >=63 | kgf/mm2 |
| **2** | Limit Stress | 27 | kgf/mm2 |
| **3** | Poisson Ratio | 0.30 | |
| **4** | Surface stress | 5000 | kgf/cm2 |
| **5** | Bending stress | 1350 | kgf/cm2 |

### 4.3 Material Properties of Steel (15Ni2Cr1Mo28)

**Table 3**

| Sl No | Description | Value | Unit |
|---|---|---|---|
| **1** | Tensile strength | >=90 | kgf/mm2 |
| **2** | Limit Stress | 55 | kgf/mm2 |
| **3** | Poisson Ratio | 0.31 | |
| **4** | Surface stress | 9500 | kgf/cm2 |
| **5** | Bending stress | 3000 | kgf/cm2 |

## 5. Theoretical Calculations for Selected Materials

### 5.1 Calculation for Cast iron (Grade 35)

**Step 1:** [D B represents PSG DDB ]

Velocity ratio(i)= 2.5

**Step 2-** Material selection:

Cast iron (Grade 35)

**Step 3**- Gear life:

L=10,000 hrs

In terms No of cycles

N = 10000*60*rpm

N = 54*10^7 cycles.

**Step 4 - Design torque:**

PSG DDB pg no 8.15

**Design Twisting Moment, [$M_t$]**

**[$M_t$] = $M_t k_d k$** **(** Initially assume for symmetric scheme $k_d k = 1.3$ )

**$M_t$ = 7160 * hp/n**

**$M_t$ = 97420 * kW/n** ($M_t$ = nominal twisting moment)

Therefore we easily find **[Mt] = 3166.15 kgf-cm.**

**Step 5- Design Stresses**

Design Stress for both Bending and Compressive strength from pg no 8.5.

**CAST IRON (GRADE 35)**

**Design Stress [$\sigma_b$], kgf/cm$^2$**

For Module(m) upto 6mm **[$\sigma_b$]=1400 kgf/cm$^2$ is** selected

For Module(m) 7mm to 10mm **[$\sigma_b$]=1350 kgf/cm$^2$ is** selected

**Compressive Stress [$\sigma_c$], kgf/cm$^2$**

**[$\sigma_c$]=5000 kgf/cm^2** is selected.

**Step 6- Calculation of centre distance between pinion and wheel:**

From Pg no 8.13.

**SPUR GEAR**

$$[\, a \geq (i+1)\ \sqrt[3]{(0.7 \div [\sigma c]^2)\, E [M_t] \div i\, \psi}\,]$$

E = 1.7 * 10^6 Kgf/Cm$^2$.

Pressure angle=0.3 radian

Compressive stress=9500 kgf/cm^2

i = 2.5

[Mt]= 3166.15 kgf-cm

Therefore Centre distance, a=14.41 cm.

**Step 7-Calculation of No of teeth(Z):**

No of teeth on Pinion

Z1= 18 teeth

No of teeth on Wheel

D.B pg no 8.1

**i (Gear ratio) = Z2/Z1 > 1,** Standard gear ratio

**Z is Number of teeth**,

Z1 on Pinion & Z2 on Wheel

Therefore Z2=45 teeth.

**Step 8-Calculation of module:**

D.B pg no 8.22

**Module(m) = 2a/Z1+Z2**

Units are in mm

Therefore m=4.2 mm, from D.B pg no 2

m=5mm is standard value.

For module from 1 to 6 mm

(D.B pg no 8.5) Bending stress = 800 kgf/cm^2 is chosen.

**Step 9-Revision of centre of distance(a):**

From D.B pg no 8.22

**Centre distance (a) = m(Z1+Z2/2)**

Units are in mm

Therefore**, a=157.5mm**.

**Step 10-Revision of Design Torque [Mt]:**

Mt=3166.15 kgf-cm => from step 4

From D.B pg no 8.1

$[\Psi = b \div a$

$b = \psi * a]$

From the above table 10 formula we calculate the face width (b)

**b=47.1mm**

**Pitch Diameter or Reference Diameter (d)**

$[d1 = m\ Z1$

$d2 = m\ Z2]$

Therefore **d1=90mm**

From D B Pg no 8.15, Pitch Line Velocity in m/s,

**Pitch Line Velocity,**

$$[V = (\pi\, d_1\, n_1) \div (60 * 1000)\ m/s]$$

**V=4.24 m/s** upto 8m/s in Table 15 D.B pg no 8.16

**kd=1.55**

To find k=> Load concentration factor,

D B Pg no 8.15, Table 14 ,

$[\ \psi_p = b \div d_1\ ]$

Therefore **k=1.03,** Now revised Design Torque(D.B pg no 8.15)

[Mt]=Mt * kd* k

**[Mt]=5054.75 kgf-cm.**

**Step 11-Check for bending stress:**

D.B pg no 8.13A

**SPUR,**

$$[\ \sigma_b = 0.7(\, i \pm 1 \div a\, b\, m_n\, y_v)\ [M_t] \leq [\sigma_b]\ ]$$

By the calculation, **bending stress=1269.21 kgf/cm^2**

The above value is greater than **Design bending stress**=**800 kgf/cm^2**.

Design is unsafe for above calculation.

**Step 12-Check for contact Compressive stress:**

D.B pg no 8.13

**SPUR**

$$[\ \sigma_c = 0.74\, i \pm 1 \div (\sqrt{(\, i \pm 1 \div i\, b)\, E\, [M_t]}) \leq [\sigma_c]\ ]$$

By the calculation, **Compressive stress= 8244.77 kgf/cm^2**

The above value is less than **design compressive stress = 7500 kgf/cm^2.**

Design is unsafe for above calculation.

[NOTE: ALL THE PSG DESIGN DATA BOOK FORMULAES ARE ADDED IN THE ABOVE CALCULATION, HERE AFTER WE ARE GOING TO ADD ONLY PAGE NUMBERS]

**5.2 Calculation for C45 Steel**

**Step 1-**

Velocity ratio(i)= 2.5

**Step 2**- Material selection:

**C45 STEEL**

**Step 3**- Gear life:

L=10,000 hrs

In terms No of cycles

N = 10000*60*rpm

N = 54*10^7 cycles.

**Step 4- Design torque:**

PSG DDB pg no 8.15

Therefore we easily find **[Mt]=3166.15 kgf-cm.**

**Step 5- Design Stresses:**

Design for both Bending and Compressive strength from PSG DDB pg no 8.5.

**Step 6- Calculation of centre distance between Pinion and Wheel:**

E = $2.15 * 10^6$ kgf/$Cm^2$.

Pressure angle=0.3 radian.

Compressive stress=9500 kgf/cm^2.

i = 2.5.

[Mt]= 3166.15 kgf-cm.

Therefore **Centre distance, a=20.26 cm.**

**Step 7-Calculation of No of teeth(Z):**

No of teeth on Pinion,

**Z1= 18 teeth**

No of teeth on Wheel,

D B Pg no 8.1

Therefore **Z2=45 teeth**

**Step 8-Calculation of module:**

D B pg no 8.22

Therefore m=4.2 mm, from D B pg no,

**m=8mm** is standard value.

For module from 7 to 10 mm

(D.B pg no 8.5)

**Bending stress = 1350 kgf/cm^2** is chosen.

**Step 9-Revision of centre of distance(a):**

From D B pg no 8.22,

Therefore**, a=252 mm**.

**Step 10-Revision of Design Torque[Mt]:**

Mt=3166.15 kgf-cm => from step 4

From D B Pg no 8.1,

From the above table 10 formula we calculate the face width (b)

b=7.56cm

Pitch Diameter

Therefore **d1=144mm**

From D B p g no 8.15, Pitch Line Velocity in m/s

**V=6.78 m/s** upto 8m/s in Table 15 D B pg no 8.16.

**kd=1.55**

To find k => Load concentration factor,

D B Pg no 8.15, Table 14

Therefore k=1.03, Now revised Design Torque (D.B pg no 8.15)

[Mt]=Mt * kd* k

**[Mt]=5054.75 kgf-cm.**

**Step 11-Check for bending stress:**

D B pg no 8.13A

By the calculation, **bending stress=307.90 kgf/cm^2**

The above value is less than Design bending stress = 1350 kgf/$cm^2$.

Design is safe for above calculation.

**Step 12-Check for contact Compressive stress:**

D B pg no 8.13

By the calculation, **Compressive stress= 4522.33 kgf/cm^2**

The above value is less than **design compressive stress = 5000 kgf/cm$^2$.**

Design is safe for above calculation.

**5.3 Calculation for 15Ni 2 Cr 1 Mo 15 Steel:**

**Step 1-**

Velocity ratio(i)= 2.5

**Step 2**- Material selection:

C45 STEEL

**Step 3**- Gear life:

L=10,000 hrs

In terms No of cycles

N = 10000*60*rpm,

N = 54*10^7 cycles.

**Step 4 - Design torque:**

PSG DDB pg no 8.15

Therefore we easily find **[Mt]=3166.15 kgf-cm.**

**Step 5- Design Stresses:**

Design for both Bending and Compressive strength from PSG DDB pg no 8.5.

**Step 6- Calculation of centre distance between Pinion and Wheel:**

E = 2.15*10^6 kgf/Cm$^2$.

Pressure angle=0.3 radian.

Compressive stress=9500 kgf/cm^2.

i = 2.5.

[Mt]= 3166.15 kgf-cm

Therefore **Centre distance, a=13.34 cm.**

**Step 7-Calculation of No of teeth(Z):**

No of teeth on Pinion,

**Z1= 18 teeth**

No of teeth on Wheel,

D B Pg no 8.1

Therefore **Z2=45 teeth.**

**Step 8-Calculation of module:**

D.B pg no 8.22

Therefore m=4.2 mm, from D.B pg no

**m=5mm** is standard value.

For module from 1 to 6 mm

(D.B pg no 8.5)

**Bending stress = 3200 kgf/cm^2** is chosen.

**Step 9-Revision of centre of distance(a):**

From D.B pg no 8.22

Therefore**, a=157.5mm**.

**Step 10-Revision of Design Torque[Mt]:**

Mt=3166.15 kgf-cm => from step 4

From D.B pg no 8.1

From the above table 10 formula we calculate the face width (b)

b=47.1mm

Pitch Diameter

Therefore **d1=90mm**

From D B Pg no 8.15,

Pitch Line Velocity in m/s,

**V=4.24 m/s** upto 8m/s in Table 15 D.B pg no 8.16

**kd=1.55**

To find k=> Load concentration factor,

D B pg no 8.15, Table 14

Therefore k=1.03, Now revised Design Torque(D.B pg no 8.15)

[Mt]=Mt * kd* k

**[Mt]=5054.75 kgf-cm.**

**Step 11-Check for bending stress:**

D.B pg no 8.13A

By the calculation, **bending stress=1269.94 kgf/cm^2**

The above value is less than Design bending stress = 3200 kgf/cm$^2$.

Design is safe for above calculation.

**Step 12 - Check for contact Compressive stress:**

D.B pg no 8.13

By the calculation, **Compressive stress= 9320.62 kgf/cm^2**

The above value is less than **design compressive stress = 9500 kgf/cm^2.**

Design is safe for above calculation.

## 6. Meshing

Meshing is one of the crucial step in the solid works design analysis. The automatic mesher in the software generates global element size and local mesh control options. The meshing model has been shown in the fig.1.

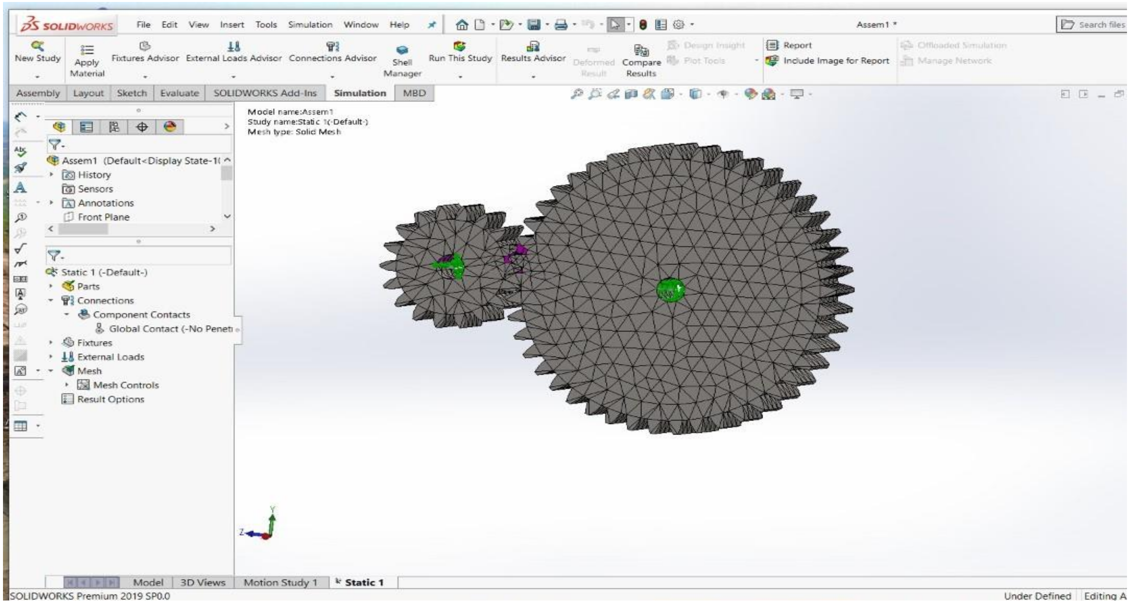


**Figure 1.** Meshing model

## 7. Design Specifications

The gear assembly was designed on machine SOLIDWORKS software using the sketcher module and other assembling options. Same gear profiles were used for gear and pinion to maximise their contact areas while performing part assembly. The gear specifications like face width, gear module, number of teeth, pitch circle diameter, pitch line velocity and pressure angle were considered and the spur gear was designed.

To include standard spur gears in Tool box:

- From the Windows (any versions) Start menu, click All Programs > SOLIDWORKS (any versions after 2014) > SOLIDWORKS Tools > Tool box Settings.
- From the Configure Hardware page, click GB box, Power Transmission and then click Gears.
- Select the required spur gears and clear all the unnecessary ones in the tool box.
- Finally close the Tool box Settings that appearing on the screen.

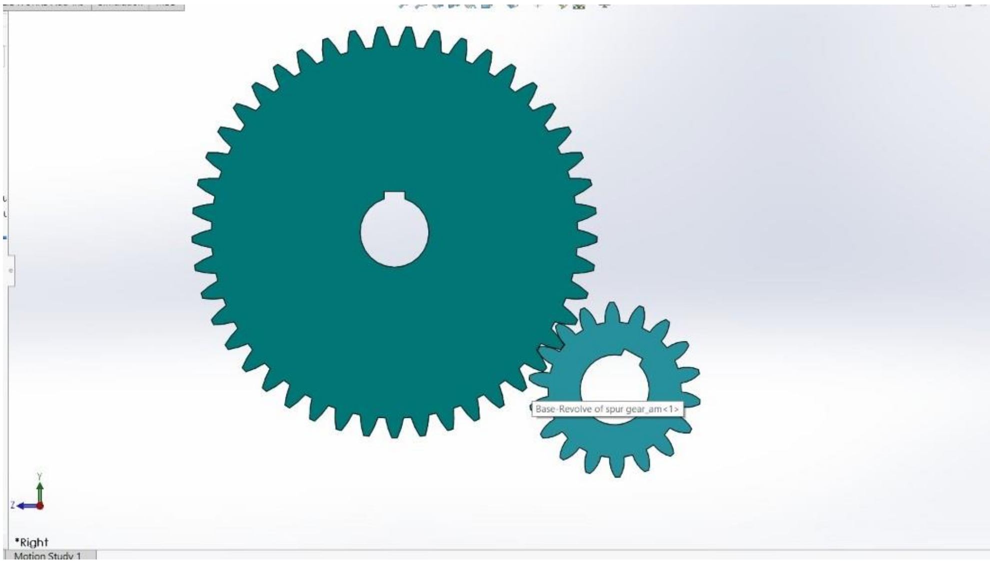


**Figure 2.** The Gear model

## 8. Results and Discussion

All the three materials are designed and simulated by Solid works simulation method. Already the Compressive and Bending stresses are calculated theoretically. Now those stresses are going to compare with the von-mises value. After that we easily find out the material which is better to that specified power transmission.

### 8.1 Cast Iron (Grade 35) Stress Simulation

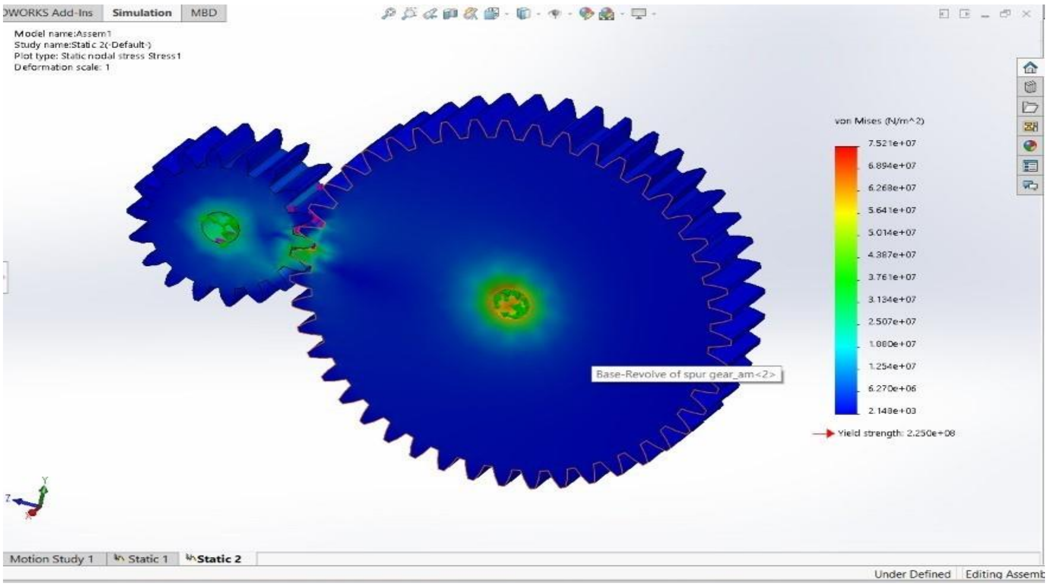


**Figure 3**

### 8.2 C45 Stress Simulation

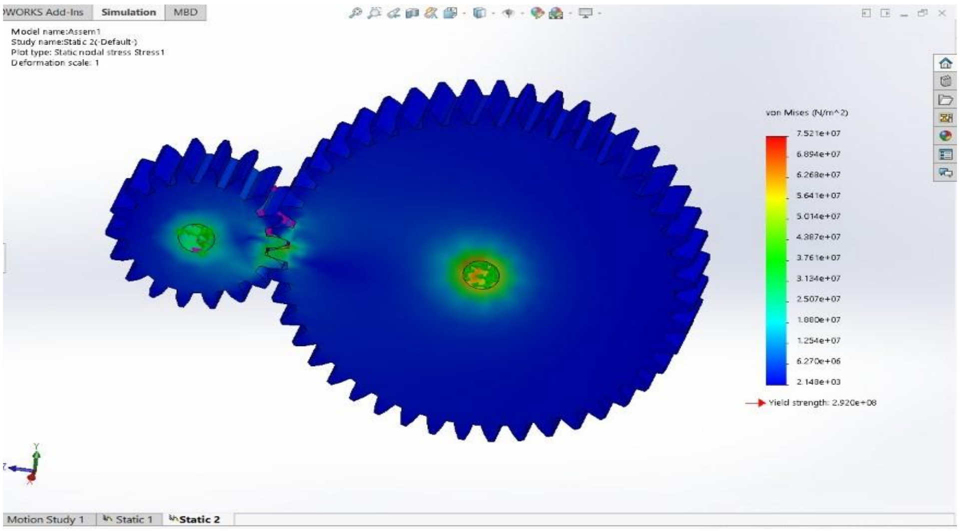


**Figure 4**

**8.3 15Ni 2Cr 1 Mo15 Steel**

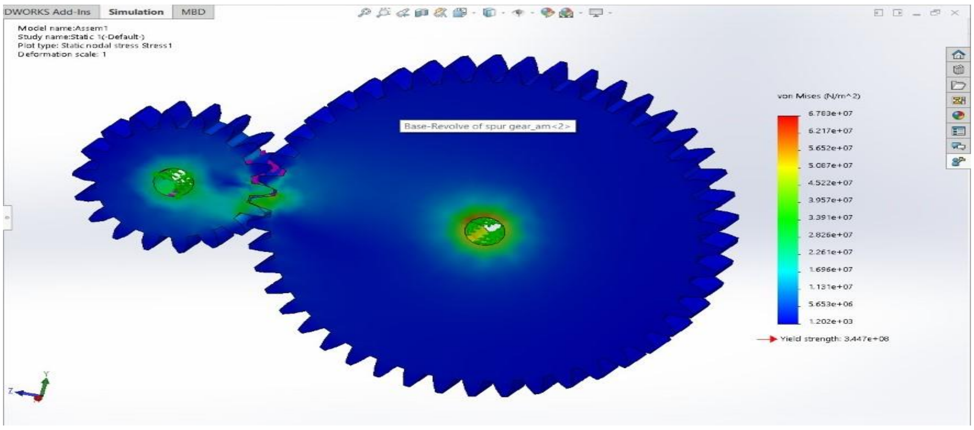


**Figure 5**

**Comparison of theoretical results with solid works data (Bending stress)**
**Table 4**

| Gear Materials | Design Bending Stress limit | Calculated value | Solid works von mises | Design Status |
|---|---|---|---|---|
| **Grade 35 Cast iron** | 800 kgf/cm2 | 1269.21 kgf/cm2 | 1268.77 kgf/cm2 | Design is unsafe. |
| **C45 steel** | 1350 kgf/cm2 | 307.90 kgf/cm2 | 1278.72 kgf/cm2 | The value is so close, it is better to skip this material. |
| **15Ni 2Cr 1 Mo 15 Steel** | 3200 kgf/cm2 | 1269.94 kgf/cm2 | 1159.41 kgf/cm2 | Design is safe. |

**Comparison of theoretical results with solid works data (Compressive stress)**
**Table 5**

| Gear Materials | Design Compressive Stress limit | Calculated value | Solid works von mises | Design Status |
|---|---|---|---|---|

| | | | | |
|---|---|---|---|---|
| **Grade 35 Cast iron** | 7500 kgf/cm2 | 8244.77 kgf/cm2 | 7659.99 kgf/cm2 | Design is unsafe. |
| **C45 steel** | 5000 kgf/cm2 | 4522.33 kgf/cm2 | 7669.92 kgf/cm2 | Design is unsafe. |
| **15Ni 2Cr 1 Mo 15 Steel** | 9500 kgf/cm2 | 9320.62 kgf/cm2 | 6916.73 kgf/cm2 | Design is safe. |

## 9. Conclusion

All the three Gear materials are theoretically calculated and analysed by using solid works simulation. From the above results it is cleared that 15Ni2 Cr 1 Mo 15 Steel is extremely good and safe when compared with other materials. For that specified data that nickel chromium steel is used for power transmission in industries.

Bending stress, friction, sliding contact, wear and tear are the main problems that are causing failures in spur gears. The recent development of FEA model of spur gears model assembly are done to simulate the compressive stress and calculate the bending stress by using von-mises option the solid works simulation. Also, that PSG DDB values and formulae's is used for calculating the bending and compressive stress mathematically in the meshing pair of spur gears.

All the values are compared with each other, if any one of the value is higher than PSG DDB stress limit value that particular material is consider to be as a failure model. Hence, we conclude that nickel Chromium is consider to be safe model it comes under the stress limit value in both mathematical and analytical values.

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